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UNITED STATES
DEPARTMENT OF THE INTERIOR
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HELIUM ACTIVITY
HELIUM RESEARCH CENTER

INTERNAL REPORT

DESIGN OF THE JACKETED BOMBS FOR THE ISOTHERMAL BURNETT APPARATUS

BY

John E. Miller

BRANCH Fundamental Research

PROJECT NO. 4330

DATE January 1965

AMARILLO, TEXAS

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ABSTRACT

This report contains the general design considerations, the principal stresses, pressures to start plastic deformation, burst pressures, and a comparison of the actual design to what would be required under the ASME code for the jacketed bombs used in the isothermal Burnett apparatus.

INTRODUCTION

The only high-pressure component of the isothermal Burnett apparatus not available from a commercial source (except by a special contract covering design and construction) was the bombs or cells for V_1 and V_2. Therefore, it was decided that it would be better to design the bombs here and have them constructed in our Machine Shop.

It was desired to have pressure jackets around V_1 and V_2 because this type of construction allows the most flexibility in operating

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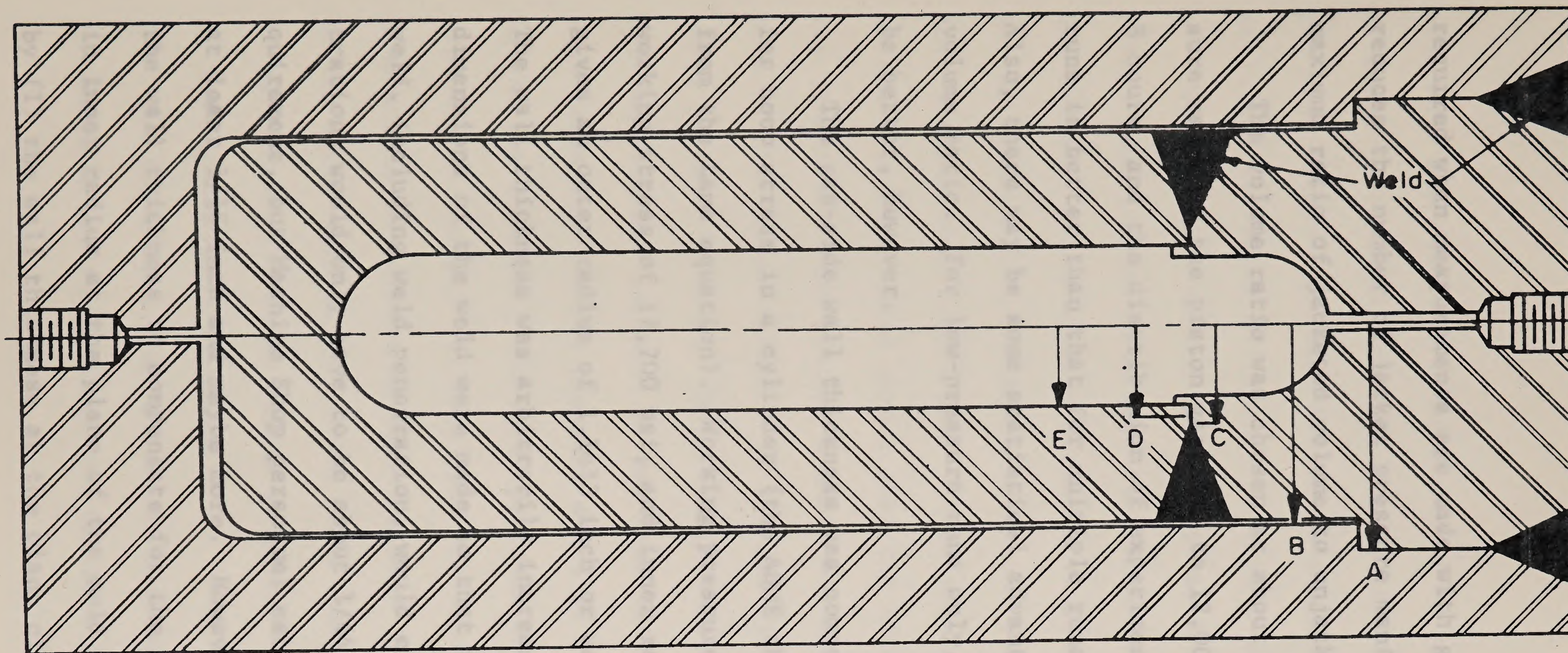
procedure. The jackets would also make it possible to check the usual methods of correcting for volume change due to elastic distortion.

The design for the inner part of the bombs (the gas cylinders) is within about 2 percent of ASME code requirements, even though the ASME code is intended for vessels with internal diameter greater than six inches and working pressure less than 3,000 psi. Internal diameter of a gas cylinder is 1 inch; working pressures is 12,000 psi (see fig. 1). The minimum wall thickness of the jacket is such that the working stress exceeds the maximum allowed by the ASME code by 61 percent. The ASME code design is based on circumferential stress and ultimate tensile strength. For thick wall cylinders made of ductile materials, the critical stress is the shear stress and failure is determined by the yield strength rather than the ultimate tensile strength. The shear stress in the gas cylinder is 40 percent of that required to start plastic deformation; the jacket is operated at 50 percent of the shear stress required to start plastic deformation.

GENERAL DESIGN CONSIDERATIONS

Size considerations were that 4" O.D. x 10" cylinders were as large as available space in the water bath would permit and still have room for the diaphragm cell.

It was planned that pressure measurements would be made with gas in both V_1 and V_2 because this procedure eliminates several steps



Dimensions at Point	Inside Radius	Outside Radius
A	1.4375"	2.000"
B	1.2656"	2.000"
C	0.625"	1.250"
D	0.5625"	1.250"
E	0.500"	1.250"

FIGURE 1. - Cross Section of one of the Jacketed Bombs

required when measurements are made with gas in V_1 only. It also reduces the number of jacket pressure manipulations and gives the maximum ratio of jacketed volume to unjacketed volume.

The volume ratio was chosen as about 2 because the entire pressure range of the piston gage (30 to 12,000 psi) could be covered in 8 hours, and the distribtuion of experimental points for multiple runs is better than that for multiple runs with a lower volume ratio. Also, there may be some statistical advantages for having a high volume ratio. For low-pressure runs only, a lower volume ratio would be better, however.

The gas-side wall thickness was computed with the Lamé equation for hoop stress in a cylinder (the ASME thick wall equation is derived from the Lamé equation). Working pressure was taken as 12,000 psi, working stress of 17,700 psi, and inner radius of 0.5 inch. This gives an outer radius of 1.1413 inch or wall thickness of 0.641 inch. The wall thickness was arbitrarily increased to 0.75 inch and the dimensions of the weld were made so that the wall thickness at the weld, including weld penetration, would exceed 0.641 inch. Weld penetration, would only have to be about 1/64 inch to meet the stated requirement, but Machine Shop personnel estimate that the weld penetrates at least 1/16 inch and maybe more. However, no allowance was made to the wall thickness to compensate for the small increase (1/16 inch) in inner radius at the plane of the weld vertex. This can be justified by (1) the wall thickness at the plane of the weld vertex has adequate strength, even if there had not been any weld penetration; (2) the gas

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side wall and weld can be prestressed by jacket pressure so that stresses in the weld will not exceed 12,000 psi; (3) the gas cylinder can only deform 2 percent before it comes in contact with the jacket wall; and (4) joint efficiency is unity, according to the ASME code.

With respect to the jacket, it was desired to have the outer radius 2.00 inches, 1/64 inch clearance between the jacket wall and gas cylinder wall, working pressure of about 10,000 psi, hoop stress at the minimum wall thickness below the yield stress, and burst pressure greater than 12,000 psi. These requirements are based, obviously, on the problem at hand rather than the ASME code. The reason is that the jacket contains oil, so a hazardous situation would not be created even if the jacket failed. If the gas cylinder should fail, it is safer to have jackets. Three things indicate that the jacket is adequate for its intended purpose: it has been tested at 15,000 psi, which is 1.65 times its working pressure and 1.25 times the working pressure of the gas cylinder; its burst pressure is over twice the working pressure of the gas cylinder; and the shear stress at the working pressure is 50 percent of that required to start plastic deformation at the inner wall of the jacket neck (the gas cylinder is operated at 40 percent of the shear stress required to start plastic deformation at its minimum wall thickness).

The bombs were constructed of free-machining 303 stainless steel to minimize galling between the gas cylinder and the inlet gas fitting.

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The bombs were constructed of free-machining 303 stainless steel to minimize galling between the gas cylinder and the inlet gas fitting.

PRINCIPAL STRESSES DUE TO PRESSURE

The five dimensions of interest are shown in figure 1, a cross-section drawing of the bomb. Dimensions at A are for the jacket neck, B for the jacket wall, C for the gas cylinder weld vertex (no penetration), D for the gas cylinder weld vertex (1/16" penetration), and E for the gas cylinder wall. All of the principal stresses, the shear stress, pressure to start plastic deformation, and burst pressures have been computed for these five dimensions. Principal stresses were computed with the Lamé equations. All equations are from Strength of Materials, Part I and II, by Timoshenko, D. van. Nostrand Co., Inc., third edition, 1956.

Nomenclature is:

σ_T = tangential, hoop, or circumferential stress

σ_{radial} = radial stress

σ_A = axial stres

θ = shear stress at a

a = inner radius

b = outer radius

r = some point between a and b

P_g = gas pressure

P_j = jacket pressure

All stresses and pressures are in psi, all dimensions are in inches.

$$\sigma_T \text{ at } r = \frac{a^2 P_g - b^2 P_j}{b^2 - a^2} + \frac{(P_g - P_j)\, a^2 b^2}{(b^2 - a^2) r^2}, \text{ Part II page 208} \qquad (1)$$

$$\sigma_{radial} \text{ at } r = \frac{a^2P_g - b^2P_j}{b^2 - a^2} - \frac{(P_g - P_j)a^2b^2}{(b^2 - a^2)r^2}; \text{ Part II page 208} \quad (2)$$

$$\sigma_A = \frac{a^2P_g - b^2P_j}{b^2 - a^2}; \text{ derived from principles stated in Part I, page 45} \quad (3)$$

$$\theta = \frac{\sigma_T - \sigma_{radial}}{2} = \frac{b^2P_g}{b^2 - a^2}; \text{ Part II page 386} \quad (4)$$

$$\text{Burst Pressure} = -2(\text{yield stress}) \ln \frac{a}{b}; \text{ Part II page 388} \quad (5)$$

For convenience, often-used functions of the 5 principal dimensions are given in table 1.

TABLE 1. - Functions of the dimensions

a	b	a^2	b^2	$b^2 + a^2$	$b^2 - a^2$	$\frac{b^2 + a^2}{b^2 - a^2}$	$\frac{a}{b}$	$\ln \frac{a}{b}$
0.5000	1.25	0.2500	1.5625	1.8125	1.3125	1.3810	.4000	-0.9163
.5625	1.25	.3164	1.5625	1.8789	1.2461	1.5078	.4500	- .7985
.6250	1.25	.3906	1.5625	1.9531	1.1719	1.6666	.5000	- .6932
1.2656	2.00	1.6017	4.0000	5.6017	2.3983	2.3357	.6328	- .4576
1.4375	2.00	2.0664	4.0000	6.0664	1.9336	3.1374	.7188	- .3303

Equations for tangential stress and shear stress are given in table 2.

TABLE 2. - Equations for tangential stress and shear stress

Dimensions of:	σ_T, psi	θ, psi
gas side wall	$1.3810\ P_g - 2.3810\ P_j$	$1.1905\ (P_g - P_j)$
gas side wall, 1/16" weld penetration	$1.5078\ P_g - 2.5078\ P_j$	$1.2539\ (P_g - P_j)$
gas side, no weld penetration	$1.6666\ P_g - 2.6666\ P_j$	$1.3333\ (P_g - P_j)$
jacket wall	$2.3357\ P_j$	$1.6666\ P_j$
jacket neck	$3.1374\ P_j$	$2.0687\ P_j$
Ruska 1/8" tube	$1.5022\ P_g$	$1.2511\ P_g$
Ruska 3/16" tube	$2.3876\ P_g$	$1.6938\ P_g$
Aminco 9/16" tube	$1.8926\ P_g$	$1.4463\ P_g$

The principal stresses; σ_T, σ_{radial}, and σ_A; and the shear stresses are listed in tables 3 and 4. These values are given at the working pressures and the hydrostatic test pressures. Inspection of the shear stress column shows that neither the gas cylinder or the jacket has been subjected to enough pressure to cause plastic deformation, which begins when the shear stress equals the yield stress (38,000 psi for 303 stainless steel).

Pressure to start plastic deformation and burst pressures are given in table 5. Burst pressures based on σ_T equal to ultimate tensile strength have been computed to show that conclusions based on this criterion can be completely misleading. One might conclude that the burst pressure at -10° F is about twice as high as it is at 150° F. This is not true because the burst pressure for annealed stainless steel, a ductile material, depends on yield strength and not ultimate tensile strength. In this case, burst pressures based on ultimate tensile strength can be in error by -19,000 psi to +34,000 psi.

Table 6 contains various information of interest for the three types of tubing used in the isothermal Burnett apparatus: Ruska 1/8" and 3/16" and American Instrument Co. 9/16" tube. The Pressure Products tubing is included because they give average experimental burst pressures. Burst pressures computed with Timoshenko's equation are within 6 percent (average) of the observed values. Burst pressures based on ultimate tensile strength would be in error by -17 percent to 87 percent.

TABLE 3. - Stresses at the working pressures[1/]

a, in.	b, in.	P_{gas}	P_{jacket}	σ_T at a	σ_A	σ_{radial} at a	θ at a
.5000[2/]	1.250	12,000	0	16,570	2,290	-12,000	14,290
.5000	1.250	12,000	9,100	- 5,024	- 8,540	-12,000	3,450
.5000	1.250	0	9,100	-21,670	-10,830	0	-10,830
.5625[3/]	1.250	12,000	0	18,090	3,050	-12,000	15,050
.5625	1.250	12,000	9,100	- 4,730	- 8,360	-12,000	3,640
.5625	1.250	0	9,100	-22,820	-11,410	0	-11,410
.6250[4/]	1.250	12,000	0	20,000	4,000	-12,000	16,000
.6250	1.250	12,000	9,100	- 4,190	- 8,130	-12,000	3,870
.6250	1.250	0	9,100	-24,270	-12,130	0	-12,130
1.2656[5/]	2.000	--	9,100	21,260	6,080	- 9,100	15,180
1.4375[6/]	2.000	--	9,100	28,550	9,725	- 9,100	18,830

1/ All pressures and stresses are in psi. Positive stresses indicate tension; negative stresses indicate compression.

2/ These dimensions are for the gas cylinder wall.

3/ These dimensions are for the case of 1/16" weld penetration.

4/ These dimensions are for the gas side weld vertex (no weld penetration).

5/ These dimensions are for the jacket wall.

6/ These dimensions are for the jacket neck.

TABLE 4. - Stresses during hydrostatic tests[1/]

a, in.	b, in.	P_{gas}	P_{jacket}	σ_T at a	σ_A	σ_{radial} at a	θ at a
.5000	1.250	17,000	0	23,480	3,240	17,000	20,240
.5000	1.250	0	15,000	-35,715	-17,860	0	-17,860
.5000	1.250	15,000	15,000	-15,000	-15,000	-15,000	0
.5625	1.250	17,000	0	25,630	4,320	-17,000	-21,320
.5625	1.250	0	15,000	-37,620	-18,810	0	-18,810
.5625	1.250	15,000	15,000	-15,000	-15,000	-15,000	0
.6250	1.250	17,000	0	28,330	5,670	-17,000	22,670
.6250	1.250	0	15,000	-40,000	-20,000	0	-20,000
.6250	1.250	15,000	15,000	-15,000	-15,000	-15,000	0
1.2656	2.000	--	15,000	35,040	10,020	-15,000	25,000
1.4375	2.000	--	15,000	47,060	16,030	-15,000	31,030

1/ All pressures and stresses are in psi. Positive stresses indicate tension; negative stresses indicate compression. Hydrostatic pressures were produced with oil.

TABLE 5. - Pressures required to start plastic deformation and burst pressures

a	b	P_{jacket} psi	Pressure required to produce the assumed stress, psi: σ_T = tensile strength at a[1/]			θ = yield[2/] stress at a	Burst[3/] Pressure
			TEMPERATURE, DEGREES F.				
			-10	80	150	-10 to 150	-10 to 150
.5000	1.250	0	103,500	69,500	50,700	31,900	69,600
.5000	1.250	9,070	119,200	85,100	66,300	41,000	
.5000	1.250	15,000	129,400	95,400	76,500	46,900	
.5625	1.250	0	94,800	63,700	46,400	30,300	60,700
.5625	1.250	9,070	109,900	78,700	61,500	39,400	
.5625	1.250	15,000	119,800	88,600	71,300	45,300	
.6250	1.250	0	85,800	57,600	42,000	28,500	52,700
.6250	1.250	9,070	100,300	72,100	56,500	37,600	
.6250	1.250	15,000	110,000	81,600	66,000	43,500	
1.2656	2.000	--	61,200	41,100	30,000	22,800	34,800
1.4375	2.000	--	45,600	30,600	22,300	18,400	25,100
.5781	2.000	--					94,300
.5156	2.000	--					103,000

1/ Tensile strength is taken as 143,000 psi at -10° F; 96,000 psi at 80° F; and 70,000 psi at 150° F. The values at -10° and 80° are taken from NBS Monograph 13, page 98, for 303 annealed stainless steel. The value at 150° is taken as 4 times the allowable stress (17,700 psi) for 302 stainless steel as given in the ASME code. All of the modern failure theories are based on yield strength, however.

2/ Yield stress is 38,000 psi, NBS Monograph 13, page 98.

3/ Burst pressure = -2 (yield strength) ln a/b. See table 6 for comparison to observed values for some tubing.

TABLE 6. - Information for various types of stainless steel tubing

Tube vendor	Ruska Inst. Corp.		American Instrument Co.	Pressure Products Inc.		
Nominal tube size	1/8"	3/16"	9/16"	1/4"	3/8"	1/2"
a, inches	.028	.060	.1562	.0600	.1225	.1550
b, inches	.0625	.09375	.2812	.125	.1875	.25
σ_T/P	1.5022	2.3876	1.8926	1.5988	2.4894	2.2489
pressure rating, psi 1/	15,000	15,000	20,000	12,300	7,500	8,450
max. σ_T, psi 2/	22,530	35,800	37,850	19,660	18,670	19,000
stainless steel no.	316	316	304 (1/4 hard)	304	304	304
yield strength, psi	38,000	38,000	75,000	35,000	35,000	35,000
pressure to start plastic deformation	30,380	22,430	51,860	26,930	20,060	21,550
calculated burst pressure	61,000	33,900	88,180	51,380	29,800	33,460
observed burst 3/ pressure	--	-	--	53,000	33,000	35,000
ln a/b	-.80296	-.44629	-.58789	-.73397	-.42572	-.47804
a/b	.4480	.6400	.5555	.4800	.6533	.6200

1/ Pressure ratings quoted by the tube vendor.

2/ Under the ASME code, allowable working stress is 18,750 psi @ 100° F. The first three entries exceed this by 20%, 91%, and 102%, respectively.

3/ The observed burst pressures are given in the Pressure Products Incorporated, sales catalog. The calculated values are within an average of 6% of the observed values.

Table 7 gives a comparison of the actual dimensions, working stresses, working pressures, and hydrostatic tests to what would be required by the ASME code. For the gas cylinder, σ_T in the wall is 6 percent lower than the maximum allowed by the code; σ_T in the weld is 2 percent higher. In the jacket, σ_T in the wall exceeds that allowed under the code by 20 percent; σ_T in the jacket neck exceeds it by 61 percent. Working stress in the 1/8" and 3/16" tubing exceeds 17,700 psi by 2 percent and 62 percent, respectively. The 5/16" tube is 1/4 hard and has a yield strength of about 75,000 psi (about twice as high as that for the annealed tubing). No specific provisions are made for working stress for hardened stainless steel in the ASME code.

Actual hydrostatic test pressure for the gas cylinder is 2,100 psi less than that required by the ASME code; that for the jacket exceeds code requirements by 550 psi.

Inspection of the safety factors based on pressure to start plastic deformation and burst pressure indicates that operating the jacket at 9,070 psi is as safe as using the Ruska 3/16" tube at 12,000 psi. Safety factors, defined as ratio of ultimate tensile strength to working stress, fall in between those for plastic deformation and burst pressure; therefore, they do not serve as a guide to anything useful as far as ductile materials are concerned.

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TABLE 7. - Comparison of actual design to ASME requirements

	Gas Cylinder wall	Gas Cylinder weld	Jacket wall	Jacket neck	Tubing 1/8"	Tubing 3/16"	Tubing 5/16"
inner radius, in.	0.500	0.5625	1.2656	1.4375	.028	.060	.1562
actual outer radius, in.	1.250	1.250	2.000	2.000	.0625	.09375	.2812
ASME outer radius	1.1413	1.284	2.229	2.532	--	--	--
actual σ_T, psi	16,570	18,090	21,260	28,550	18,030	28,650	22,710
ASME σ_T 1/	17,700	17,700	17,700	17,700	17,700	17,700	17,700
working pressure	12,000	12,000	9,070	9,070	12,000	12,000	12,000
ASME W.P. 2/	12,800	11,740	7,580	5,640	11,780	7,410	9,350
hydrostatic test	17,000	17,000	15,000	15,000	17,000	17,000	17,000
ASME hydro. test 3/	19,100	19,100	14,450	14,450	19,100	19,100	19,100
P to start plastic deformation	31,900	30,300	22,800	18,400	30,380	22,430	51,860
burst pressure	69,600 4/	60,700 4/	34,800	25,100	61,000	33,900	88,180
$\frac{\text{P to start plastic deform.}}{\text{W.P.}}$	2.66	2.52	2.51	2.03	2.53	1.87	4.32
$\frac{\text{Burst P}}{\text{W.P.}}$	5.80 4/	5.06 4/	3.84	2.77	5.08	2.82	7.35
$\frac{70,800}{\sigma_T}$	4.27	3.91	3.33	2.48	3.93	2.47	3.12

1/ For maximum temperature of 150° F.

2/ For actual a and b and allowable stress = 17,700 psi @ 150° F.

3/ ASME hydrostatic test pressure = $(W.P.)(1.5)(\frac{18,750}{17,700}) = (1.589)(W.P.)$ for maximum design temp of 150° F and hydrostatic tests below 100° F.

4/ Burst pressure computed from dimensions with no allowance for the pressure jacket. Actual burst pressure would be higher.

on yield strength and not ultimate tensile strength. This would seem to be much more logical, especially at very high working pressures.

The ASME code (1959) states in Par. UW-9 that with respect to design of welded joints: (a) the types of welded joints for arc and gas welding are listed in table UW-12; (b) Welding Grooves - the dimensions and shape of the edges to be joined shall be such as to permit complete fusion and complete joint penetration. In Par. UW-12 Joint Efficiencies: (a) The joint efficiencies depend on the type of joint and on the degree of examination. In Par. UG-27 (b): E = joint efficiency for or the efficiency of appropriate joints in cylinderical shells and any joint in spherical shells, ...Where there is no joint normal to the direction of stress under consideration (e.g. no longitudinal joint when circumferential stress formulas are used), E = 1.

The ASME code recommends the following dimensions for single V butt joints or groove welds: maximum thickness to be jointed: 3/4 inch; thickness between bottom of V and bottom of plate: 1/16 ± 1/32 inch; angle between vertex and side of V: 30° to 37°. In general, welds that have been machined flush are stronger than those that have not because possible stress risers are eliminated. Welds made with stainless steel base metals should be stress-relieved or normalized.

The welds in the bombs for the Burnett apparatus were made with an electric arc; staintrode D welding eutectic was used for the filler (95,000 psi tensile strength). Three or four passes were required to deposit the filler. The welds were stress-relieved after they had been machined flush to the surface. The dimensions of the groove differ from the

dimensions recommended in that the thickness from the bottom of the groove to the bottom of the plate is 1/8" instead of 1/16"; the angle between the vertex and the side of the V is 22° instead of 30°.

THERMAL STRESSES

Any residual stresses in the cylinder produced during welding should have been dissipated by the stress-relieving step. The only other cause of high stress is that due to temperature differences across the gas cylinder wall. These could be produced when the cylinder is initially charged and after gas is expanded from V_1 into V_2. Timoshenko gives equations for computing thermal stress when there is a constant temperature difference in the cylinder wall. These equations are in Part II, page 232.

$$(\sigma_T)_{r=a} = \frac{E\alpha\,(T_i - T_o)}{2(1-\mu)\ln b/a}\left[1 - \frac{2b^2 \ln b/a}{b^2 - a^2}\right] \quad (6)$$

$$(\sigma_T)_{r=b} = \frac{E\alpha\,(T_i - T_o)}{2(1-\mu)\ln b/a}\left[1 - \frac{2a^2 \ln b/a}{b^2 - a^2}\right] \quad (7)$$

where for stainless steel: $E = 29 \times 10^6$ psi

$\alpha = 9.2 \times 10^{-6}$ in/in/°F @ room temp

T_i = temperature at a, °F

T_o = temperature at b, °F

$\mu = .305$

The thermal stress is highest at the inner wall (a) when $(T_i - T_o)$ is negative, as when the bath is at 150° F and gas is

dimensions recommended in that the thickness from the bottom of the groove to the bottom of the plate is 1/8" instead of 1/16"; the angle between the vertex and the side of the V is 22° instead of 30°.

THERMAL STRESSES

Any residual stresses in the cylinder produced during welding should have been dissipated by the stress-relieving step. The only other cause of high stress is that due to temperature differences across the gas cylinder wall. These could be produced when the cylinder is initially charged and after gas is expanded from V_1 into V_2. Timoshenko gives equations for computing thermal stress when there is a constant temperature difference in the cylinder wall. These equations are in Part II, page 232.

$$(\sigma_T)_{r=a} = \frac{E\alpha\,(T_i - T_o)}{2(1-\mu)\ \ln b/a}\left[1 - \frac{2b^2\ \ln b/a}{b^2 - a^2}\right] \qquad (6)$$

$$(\sigma_T)_{r=b} = \frac{E\alpha\,(T_i - T_o)}{2(1-\mu)\ \ln b/a}\left[1 - \frac{2a^2\ \ln b/a}{b^2 - a^2}\right] \qquad (7)$$

where for stainless steel: $E = 29 \times 10^6$ psi

$\alpha = 9.2 \times 10^{-6}$ in/in/°F @ room temp

T_i = temperature at a, °F

T_o = temperature at b, °F

$\mu = .305$

The thermal stress is highest at the inner wall (a) when $(T_i - T_o)$ is negative, as when the bath is at 150° F and gas is

admitted to the bombs at 70° F. The actual situation is that the inlet gas will usually be above room temperature and will be warmed even more before it enters V_1 and V_2. Also, the inner walls of V_1 and V_2 will be at 150° F initially; therefore, it would be impossible to have an 80° F temperature difference from a to b. Taking $(T_1 - T_o)$ as -80° F should give the maximum possible thermal stress. This turns out to be $(\sigma_T)_{r=a}$ = 19,800 psi due to thermal stress. If this is superimposed on σ_T due to pressure at 12,000 psi, the shear stress would be 24,200 psi, or 64 percent of that required to start plastic deformation.

The actual thermal stress should be a small fraction of 19,800 psi. If $(T_i - T_o)$ is positive, the thermal stress at a will be negative - indicating compression. This would tend to cancel the hoop stress caused by internal pressure.

CAUSES OF FAILURE

The isothermal Burnett apparatus was designed for use with inert gases and light hydrocarbons, but not oxygen or hydrogen. The outside of the bombs is in contact with a water - antifreeze solution and the jackets contain oil. None of these things is corrosive to stainless steel. The gas cylinder is subjected to alternating stresses (tension and compression) but the fatigue strength is 34,000 psi and the working stress plus reasonable thermal stress does not exceed the fatigue strength. The maximum operating temperature is 150° F. Stress calculations indicate that working pressures are far below that required to

start plastic deformation. The cylinders are not subjected to severe shocks. The above considerations eliminate the following common causes of failure:

plastic deformation due to excess pressure

fatigue failure due to residual stress or cold work

fatigue failure due to alternating stresses above the fatigue strength

severe impact

stress corrosion

carbide precipitation

intergranular corrosion

hydrogen embrittlement

AUTOFRETTAGE

Subjecting a thick-wall cylinder to a pressure between the pressure to start plastic deformation and the burst pressure will cause plastic deformation beginning at the inner wall; but not through the wall to the outer radius. When the cylinder is depressured, the inner wall does not return to its original position. The outer part of the wall is still in an elastic state, however, and it tries to force the inner part of the wall back to its initial position; this creates a favorable compressive stress on the inner wall. The next time the cylinder is subjected to pressure, the pressure required to start plastic deformation will be higher. This is because of the compressive stress on the inner wall and increased yield strength of the material at the inner wall. Using this procedure on a stainless steel cylinder

would possibly double the effective yield strength and burst pressure. The only serious consequence is that the pressure distortion coefficient could not be computed with certainty because of changed dimensions and mechanical properties.

Addendum to IR No. 67

"Design of the Jacketed Bombs for the Isothermal Burnett Apparatus"

By

John E. Miller

After the above report was distributed, it was discovered that an incorrect assumption was made in computing the pressure required to start plastic deformation at the inner wall of a cylinder. This pressure is usually called P_y in the recent literature.

This addendum contains the corrected values of P_y, pertaining to the jacketed bombs and the associated tubing. Also included is a comprehensive bibliography on the subject of behavior of cylinders subjected to pressure. Underlined numbers in parentheses in this addendum refer to items in the included bibliography.

The equation usually used to estimate P_y, is (9), (14), (24):

$$P_y = \frac{\sigma_{y.01\%}}{\sqrt{3}} \left(\frac{b^2 - a^2}{b^2}\right) \tag{8}$$

where $\sigma_{y.01\%}$ is the yield strength for 0.01% offset in a simple tensile test. For stainless steel, the yield strength is usually given for 0.2% offset, but one of the papers (9) in the bibliography gives experimental values for both 0.01% and 0.2% offset for several types of stainless steel. The average ratio of $\sigma_{y0.2\%}$ to $\sigma_{y0.01\%}$ for the austenitic stainless steels is 1.33; therefore, if $\sigma_{y0.2\%}$ is 38,000 psi for 303 stainless steel, the value of $\sigma_{y0.01\%}$ can be estimated as 28,500 psi. The same

procedure may be used to estimate yield strength at 0.01% offset for 304 and 316 stainless steel; 26,300 psi and 28,500 psi respectively. The recalculated values of P_y are given in table 8. The recalculated values are lower by a factor of about 2.3.

The values of P_y in table 8 should be used in preference to the following parts of IR No. 67:

1. Entries in table 5 (page 14) under column heading: "θ = yield stress at a"
2. Table 6 (page 15); the line for "pressure to start plastic deformation"
3. Table 7 (page (17); the lines for "P to start plastic deformation" and "P to start plastic deform./W.P."

Also, statements concerning the shear stress-pressure to start plastic deformation relationship on pages 4, 7, 11, and 20 should be disregarded.

On page 22, the statement is made that autofrettage could possibly double the burst pressure of a stainless steel cylinder. According to (9), (10), and (11), autofrettage (or prestressing) doesn't have much influence on burst pressure, although it is possible to double the elastic breakdown pressure.

A method for computing strain in the plastic region is given in (24); this method is used in HRC IR No. 73 to compute a bore deformation-bore pressure curve for the gas cylinder of the jacketed Burnett bomb. In this particular case, the cylinder can be subjected to a bore pressure

that is twice as high as the initial elastic breakdown pressure without causing an appreciable uncertainty in the distortion correction coefficient.

Burst pressures computed with equation (5) on page 9 are in fairly good agreement with experimental burst pressures for stainless steels (9), (Pressure Products Sales Catalog also); but it may not give good results for other kinds of metals. Another burst pressure equation, which is supposed to give burst pressures to within 15% for a wide variety of materials, is given in (9).

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May 12, 1965

TABLE 8. - Recalculated values of pressure required to start plastic deformation: P_y

Description	Stainless Steel Type No.	Estimated $\sigma_{y.01\%}$, psi	a, in	b, in	P_y[1/], psi
Gas Cylinder	303	28,500	0.500	1.2500	13,820
Jacket	303	28,500	1.2656	2.0000	9,860
Ruska 3/16" tube	316	28,500	0.0600	0.09375	9,715
Ruska 1/8" tube	316	28,500	.0280	.0625	13,150
Aminco 9/16" tube	304[2/]	56,390	.1562	.2812	22,500
Pressure Prod. 1/4" tube	304	26,300	.0600	.1250	11,690
Pressure Prod. 3/8" tube	304	26,300	.1225	.1875	8,700
Pressure Prod. 1/2" tube	304	26,300	.1550	.2500	9,350

1/ P_y is computed from equation (8).

2/ The American Instrument Company 9/16" tube is 1/4 hard.

BIBLIOGRAPHY

1. Blair, J. S. "Stresses in Tubes Due to Internal Pressure". Engineering (London), v. 170 (1950) page 218.
2. Brown, W. F., and F. C. Thomson. "Strength and Failure Characteristics of Metal Membranes in Circular Bulging." Trans. ASME, v. 71, page 575 (1949).
3. Buxton, W. J., and W. R. Burrows. "Formula for Pipe Thickness". Trans. ASME, v. 73, 1951, page 575.
4. Cooper, W. E. "The Significance of the Tensile Test to Pressure Vessel Design." Welding J., v. 36, pp. 49s-56s (1957).
5. Constantino, C. J. "The Strength of Thin-Walled Cylinders Subjected to Dynamic Internal Pressures." Trans. ASME, March 1965, pp. 104-108.
6. Constantino, C. J., M. A. Salmon, and N. A. Weil. "Effect of End Conditions on the Burst Strength of Finite Cylinders." J. Appl. Mech., March 1964, pp. 97-104.
7. Crossland, B., and J. A. Bones. "The Ultimate Strength of Thick-Walled Cylinders Subjected to Internal Pressure." Engineering (London), Jan. 21, 1955, page 80.
8. _____. Inst. Mech. Eng. (London), v. 172, pp. 777, (1958).
9. Faupel, J. H. "Yield and Bursting Characteristics of Heavy-Wall Cylinders." Trans. ASME, v. 78, 1956, pp. 1031-1064.
10. _____. "Residual Stresses in Heavy-Wall Cylinders." J. Franklin Institute, v. 259, 1955, pp. 405-419.
11. _____. "Some Considerations of the Mechanics and Design Limitations of Autofrettage." J. Franklin Institute, June 1960, pp. 474-489.

12. _____. "Designing for Shrink Fits." Machine Design, v. 26, page 114, (1954).

13. Faupel, J. H., and D. B. Harris. "Stress Concentration in Heavy-Walled Cylindrical Pressure Vessels." IEC, v. 49, No. 12, Dec. 1957, page 1979.

14. Faupel, J. H., and A. R. Furbeck. "Influence of Residual Stress on Behavior of Thick-Wall Closed-End Cylinders." Trans. ASME, v. 75, pp. 345-354 (1953).

15. Jorgensen, S. M. "Overstrain and Bursting Strength of Thick-Walled Cylinders." Trans. ASME, April 1958, pp. 561-570.

16. Love. A. E. H. "Mathematical Theory of Elasticity." Dover 4th edition (1944).

17. MacGregor, C. W., L. F. Coffin, and J. C. Fisher. "Partially Plastic Thick-Walled Tubes." J. Franklin Inst., v. 245 (1948) pp. 135-158.

18. _____. "The Plastic Flow of Thick-Walled Tubes with Large Strains." J. Applied Physics, v. 19, March 1948, pp. 291-297.

19. Manning, W. R. D. "Strength of Cylinders." IEC, v. 49, No. 12, Dec. 1957, pp. 1969-1978.

20. Ranoc, T., and F. R. Park. "On The Maximum Numerical Value of the Tangential Stress in Thick-Walled Cylinders." J. Appl. Mech., March 1953, pp. 134-137.

21. Salmon, M. A. "Plastic Instability of Cylindrical Shells With Rigid End Closures." J. Appl. Mech. Sept. 1963, pp. 401-409.

22. Schneider, R. W. "Prestressing a Two-Layer Pressure Vessel by Plastic Deformation." J. of Engineering for Industry (ASME), Feb. 1965, pp. 97-103.

23. Steele, M. C., and John Young. "An Experimental Investigation of Over-Straining in Mild Steel Thick-Walled Cylinders by Internal Fluid Pressure." Trans. ASME, v. 74 (1952), pp. 355-363.

24. Svennson, N. L. "The Bursting Pressure of Cylindrical and Spherical Vessels." J. Appl. Mech, March 1958, pp. 89-96.

25. Timoshenko, S. "Strength of Materials, Part II," D. Van Nostrand Co., Inc., 3rd edition (1956).

26. Weil, N. A. "Bursting Pressures and Safety Factors for Thin-Walled Vessels." J. Franklin Inst., Feb. 1958, pp. 97-116.

27. W. M. Kellog Co., "Design of Piping Systems," Chapters 1 and 2. Wiley, 2nd edition (June 1964).